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K. Sano

K. Mitsui

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ANALYSIS OF HERMETIC ROLLING PISTON TYPE COMPRESSOR NOISE, AND COUNTERMEASURES

Kiyoshi Sano : Chief Engineer, Matsushita Electric Industrial Co., Ltd.
Noji-cho, Kusatsu City, Shiga Pref. 525, Japan

Kimiyoshi Mitsui: Engineer, Matsushita Electric Industrial Co., Ltd.
Noji-cho, Kusatsu City, Shiga Pref. 525, Japan

ABSTRACT

In order to locate the various sources of noise of the hermetic rolling position type compressor for room air conditioners, we experimentally analyzed the compressor noise and found that a main cause of the noise was the acoustically amplified pressure pulsation in various parts of the compressor. Moreover, with reference to results of this noise source analysis, several procedures were devised for an effective reduction of pressure pulsation in various internal spaces of the compressor. As a result, the noise of the rolling piston type compressor could be reduced to a remarkable extent.

1. INTRODUCTION

The rolling piston type compressor which is widely used in room air conditioner today is a main noise source of the air conditioner. However, since the mechanism of generation of the noise and its characteristics remains to be fully elucidated, it has been an important but difficult problem to reduce the compressor noise.

This report describes some of the results of a comprehensive, detailed investigation of various noise sources within the compressor which can be experimentally confirmed. The results of the investigation showed that the noise radiated from the compressor within the frequency range up to 500 Hz is predominantly the noise generated by the solid body motion of the compressor due to the unbalanced forces and torques and unbalanced electromagnetic forces acting on the compressor. (1)

In the frequency range of 500 to 1600 Hz, the air-borne noise due to the transmission of the internal noise of the compressor through the wall of its hermetic shell is a predominant noise. In the frequency range exceeding 1600 Hz, the noise is composed of the air-borne noise and, in a predominant proportion, the solid-borne noise which is generated as the forces acting in the compressor mechanism excites a deformed shell motion. It was also found that since the solid-borne noise is governed by the vibration characteristics and vibration transfer characteristics of various parts of the compressor, the construction of the compressor is a determinant factor in

the noise profile.

2. ANALYSIS OF NOISE SOURCES

2.1 Test compressor and test procedures

The compressors used for the analysis of noise sources were those with an output of 0.55 KW to 0.95 KW, especially 0.75 KW, and a rotational speed of approximately 3450 rpm.

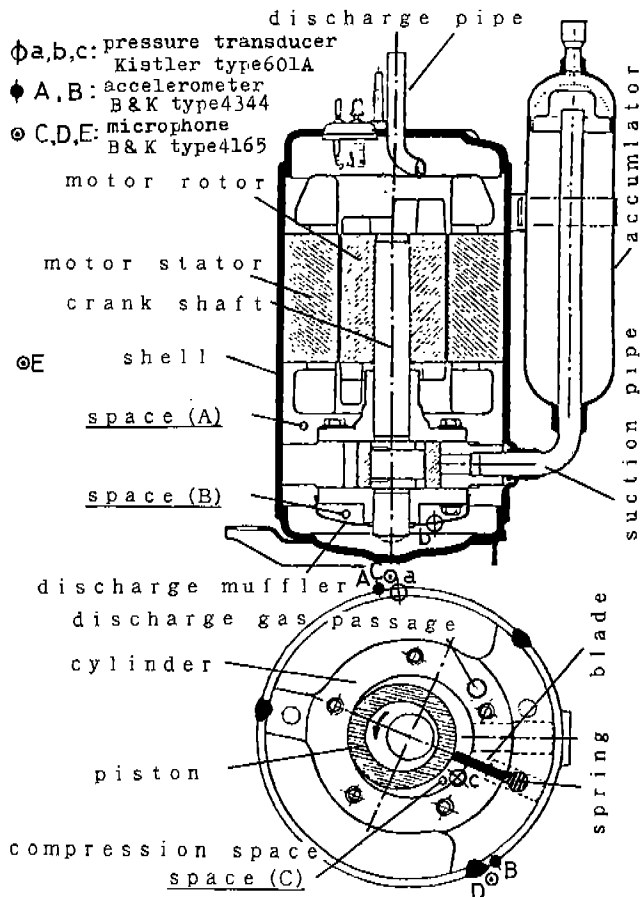


Fig. 2.1 Sectional view of test compressor

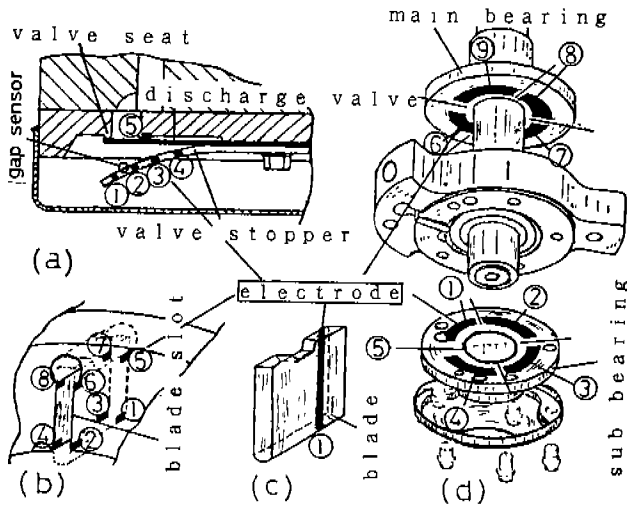


Fig. 2.2 Schematic views of electrode type experimental devices for detection of slap and slide between compressor components

Fig. 2-1 shows the construction of test hermetic rolling piston type compressors. In various parts of each compressor, sensors adapted to measure the pressure pulsations and shell vibration acceleration were set in the positions indicated in Fig. 2.2.

The noise was measured in the near position 1 cm away from the exterior wall of the shell and for 1/3 oct frequency analysis, at the position 30 cm away. Since the compressor noise and shell vibration acceleration are in a coherent relationship over substantially the entire frequency range, the shell vibration acceleration was mainly utilized for analysis.

Fig. 2.2 shows the test setup wherein electrodes (ON during contact and OFF during non-contact) were disposed in various compressor parts for detection of slap and slide between compressor mechanical parts.

Signals from the sensors were measured with various B & K instruments and the data were analyzed using FFT (HP5423A, Nicolle 660A, GR 2502). The compressor was operated at the test conditions of suction temperature $T_s = 18^\circ\text{C}$, suction pressure $P_s = 5.3 \text{ kg/cm}^2\text{G}$ and discharge pressure $P_d = 21.13 \text{ kg/cm}^2\text{G}$.

2.2 General characteristics of compressor noise

Fig. 2.3 shows the results of simultaneous measurements of shell vibration acceleration and discharge valve displacement per refrigerant cycle. The crank angle was assumed to be $\theta = 0^\circ$ when the discharge stroke has ended. Fig. 2.4 shows the result of a 3D analysis of shell vibration acceleration per cycle.

The above experimental results indicate that both the sound pressure waveform and shell vibration acceleration are at remarkably high power levels while the discharge valve is open ($\theta = 220^\circ\text{--}360^\circ$). In all the test compressors, the impact waveforms indicated at (a), (b) and (c)

were observed at substantially identical time-points, though their peak-to-peak levels varied fairly much.

These characteristics indicate that the compressor noise is composed of noises from various sources. Particularly the fact that these characteristics are at high levels while the discharge valve is open suggests strongly that the noise is generated as the compressed refrigerant gas is abruptly discharged into various spaces within the compressor.

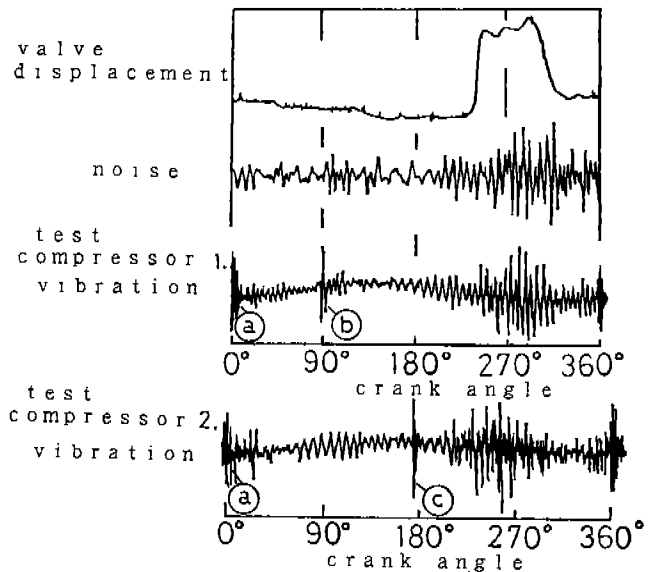


Fig. 2.3 Typical data of discharge valve displacement, shell vibration acceleration and noise

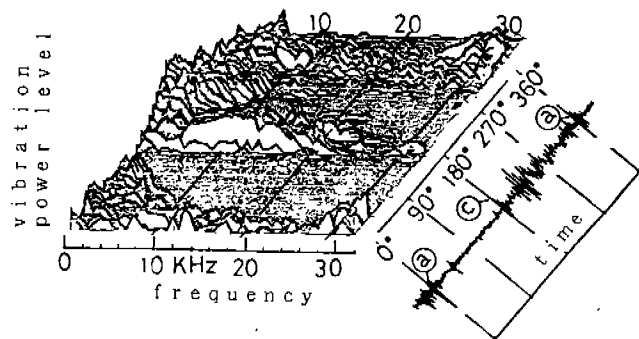


Fig. 2.4 Typical 3D analysis data of shell vibration acceleration per cycle

2.3 Relation of pressure pulsations in compressor spaces with noise

In view of the contribution of the pressure pulsation of compressed refrigerant gas to the compressor noise as suggested in the previous section, the relationship of the pressure pulsation in various compressor spaces with the compressor noise was investigated.

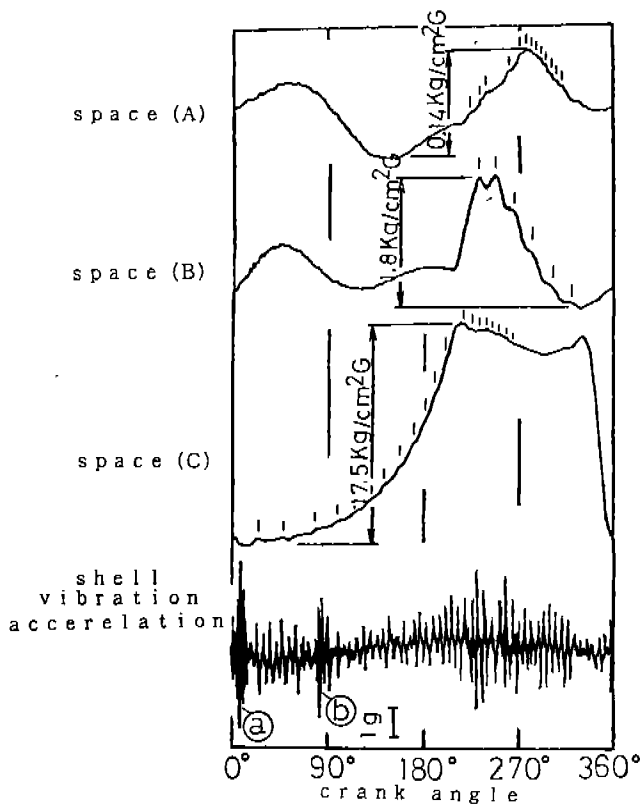


Fig. 2.5 Typical data of pressure pulsation (indicator diagram) in various compressor spaces and of shell vibration acceleration

Fig. 2.5 shows the results of simultaneous measurements of typical pressure pulsations in compressor spaces and shell vibration acceleration in a time domain. Fig. 2.6 shows the power spectra of pressure pulsations in various spaces and shell vibration acceleration and the coherent function between them.

The acoustic characteristics (converted from data generated in air) of the hermetic shell space (space A) and discharge muffler space (space B) are also shown.

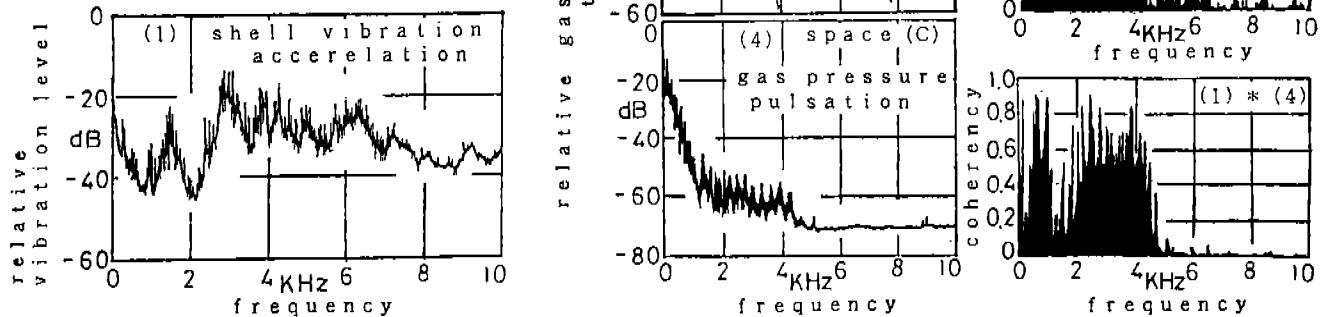


Fig. 2.6 The power spectra of pressure pulsation in various compressor spaces and shell vibration acceleration, and the coherent function between them

A detailed examination of the time domain gas pressure pulsation waveform of Fig. 2.5 indicates that a high frequency pressure pulsation indicated by the | mark is superimposed on the pressure pulsation waveform in each space. It is also observable that the high frequency pressure pulsation is excited markedly during the phase of $\theta = 220^\circ - 300^\circ$ when the discharge valve displacement is large. This coincides with the period when the shell vibration acceleration and noise acoustic pressure waveform increase.

Now, from the frequency domain analysis shown in Fig. 2.6, it appears that the peaks in the power spectra of pressure pulsations in shell internal space and discharge muffler space are in substantial agreement with the frequencies where the acoustic characteristics of the respective spaces show an acoustic characteristic which is a well-known cavity resonance. According to an estimation from the coherent function, the sum of the contributions of pressure pulsations in various compressor spaces to shell vibration acceleration is fairly large except in the frequency ranges of 200 to 400 Hz, 1600 to 2000 Hz, and over 7000 Hz. It means that the pressure pulsation in compressor spaces is a major source of compressor noise.(2) It appears that there is no definite peak-forming frequency in the power spectrum of pressure pulsation in the cylinder compression space (space C).

However, many time domain pressure pulsation data indicate that, as shown again in Fig. 2.7, a high frequency pressure pulsation is superimposed on the pressure pulsation waveform. It is still more interesting to note that the wavelength of the above pressure pulsation of a high frequency component becomes shorter as the stroke advances. Thus, as the crank angle is advanced and the cylinder compression space diminishes, the pulsation frequency tends to become higher. In order to clarify this phenomenon, the acoustic characteristic of the cylinder compression space shown in Fig. 2.1 was measured.

The results of measurements are shown in Fig. 2.8. It is seen that many resonance frequencies appear in the cylinder compression space and that with the advance of crank angle the resonance frequency corresponding to each order becomes progressively higher. (2)

Fig. 2-9 shows the change in the resonance frequency at each order as derived from results of the acoustic experiment against crank angle, wherein the frequency change of the high frequency pressure pulsation shown in Fig. 2.7 is indicated by the x mark.

The above results show that the frequency change of the pressure pulsation observed in cylinder compression space during operation of the compressor has the same trend as that of the change in resonance frequency of second order found in the acoustic experiment. Thus, in the cylinder compression space, the constant change of acoustic characteristic is followed by a change in frequency of the excited pressure pulsation. The cause of the aforementioned high frequency pressure pulsation is the re-expansion of compressed refrigerant gas remaining in the top clearance while the discharge valve is closed. (3) When the discharge valve is open, the cause may be a change in the volume density of compressed refrigerant gas due to its rapid expansion in the neighborhood of the discharge port.

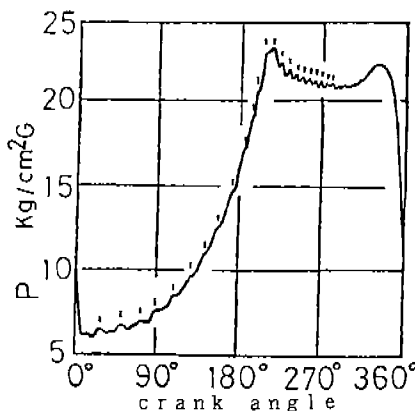


Fig. 2.7 Typical data of the high frequency component of the pressure pulsation in cylinder compression space

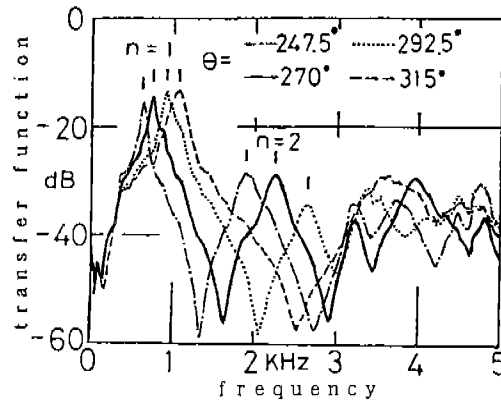


Fig. 2.8 Acoustic characteristics (converted from data in air) of cylinder compression space at various crank angles

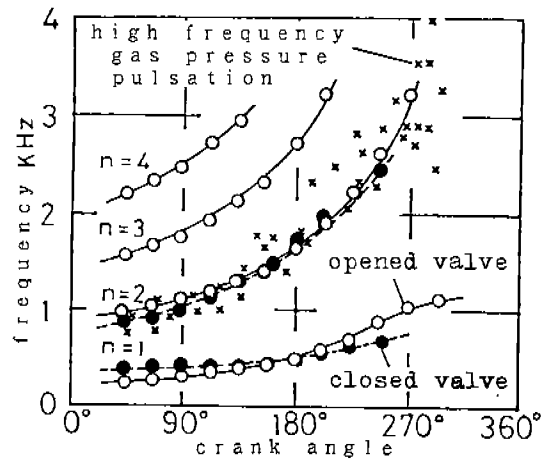


Fig. 2-9 Typical data of the change in resonance frequency of each order and the actually observed change in the high frequency component of pressure pulsation in cylinder compression space

As the compressor speed was reduced, the frequency change of the high frequency component of pressure pulsation in cylinder compression space while the discharge valve was open became such that the first-order resonance frequency change component of acoustic characteristic shown in Fig. 2.9. was also markedly excited. It was also found that when the clearances between various parts were increased, with the consequent increase of gas leaks, the above high frequency pressure pulsation was less liable to occur and consequently the noise reduced as well.

As a whole, the pressure pulsation excited and acoustically amplified in various compressor inner spaces, except at certain frequencies, is a major source of the noise radiated by the compressor.

2.4 Relation of the slap and slide between compressor mechanical members with noise

2.4.1 Slap between the discharge valve, valve seat and stopper

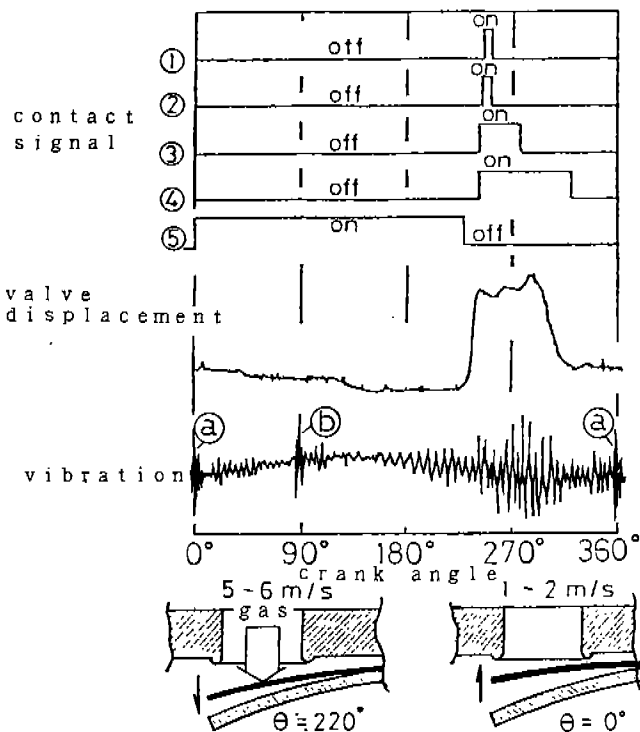


Fig. 2.10 Typical data of discharge valve motion and shell vibration acceleration waveform, and schematic views of discharge valve motion

Under the test conditions, the discharge valve is raised by compressed refrigerant gas at $\theta \approx 220^\circ$ to slap against the discharge valve stopper. As the discharge stroke ends at $\theta = 0^\circ$ or 360° , the valve slaps against the valve seat. Fig. 2.10 shows the discharge valve displacement, the change of contact state of the discharge valve against the stopper and seat, and shell vibration acceleration as measured all simultaneously.

The observed results indicate that the impact waveform (a) appearing at $\theta = 0^\circ$ in the shell vibration acceleration waveform coincides exactly with the signal from electrode 5 representing the contact of the discharge valve with the seat, so that the occurrence of slap between the discharge valve and seat at this moment can be ascertained. The slap velocity at this moment is 1 to 2 m/sec. At $\theta \approx 220^\circ$, the discharge valve slaps against the stopper at a speed of 5 to 6 m/sec. However, only a very small impact waveform is produced in the shell vibration acceleration waveform, or none is observed in many cases. This is probably because the discharge valve is raised along the contour of the stopper so that the slap is mitigated. The impulsive noise generated by these slaps of the discharge valve is scarcely audible even when the peak-to-peak level of impact waveform referred to above and shown in Fig. 2.10 is approximately 10 times as great (at transition or when the thickness of the discharge valve is small), for the excited frequency is extremely high. When a peak-to-peak level is low and the contribution of

noise from other sources is predominant, this impulsive noise plays only a negligible part in the compressor noise. (See Fig. 2.4)

2.4.2 Slap between the blade and blade slot

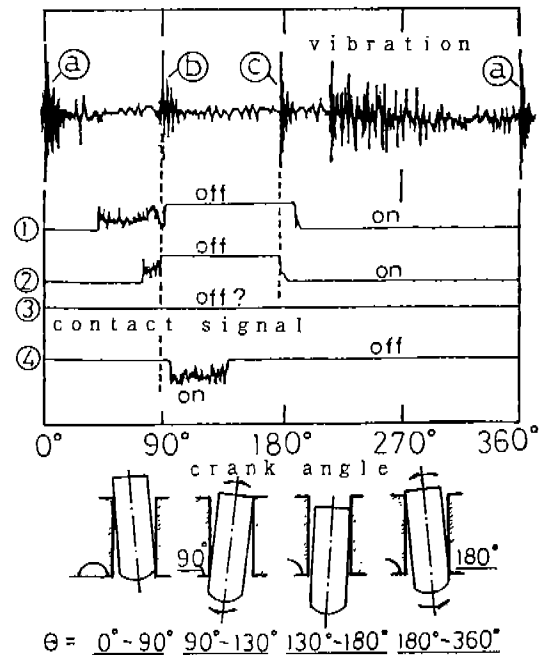


Fig. 2.11 Typical data of blade motion and shell vibration acceleration waveform, and schematic views of blade motion

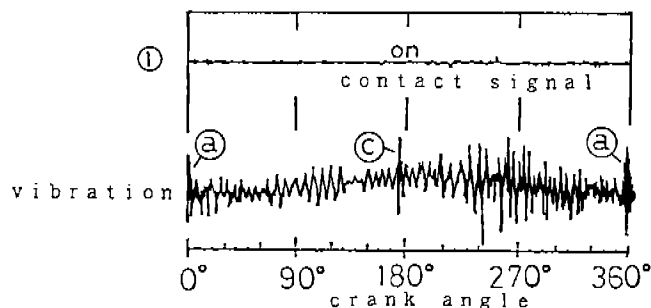
Fig. 2.11 shows the measured changes of contact between the blade and blade slot.

From these data, it is apparent that the impact waveform (b) observed at around $\theta \approx 90^\circ$ in the shell vibration acceleration waveform is attributable to the slap which occurs as the lateral side of the blade contacts the blade slot at an angle. The impact waveform (c) observed at $\theta \approx 180^\circ$ represents the slap which occurs when the nose of the blade contacts the blade slot at an angle in the opposite direction. The signals representing contact of the blade with the electrode disposed in the cylinder are in timed relation with impact waveforms (b) and (c). Thus, the contact signals followed, in perfect synchronism, advance or delay in the onset of generation of impact waveforms owing to sharp changes in pressure conditions. This is a more positive evidence that the impact waveforms (b) and (c) are attributable to the slap between the blade and blade slot. The impulsive noise of impact waveforms (b) and (c) is contributing at frequencies over 2000 Hz. Thus, the degree of its contribution to the compressor noise is only slightly greater than that of the slap between the discharge valve and the valve seat. However, this impulsive noise is audible, though faintly, in the case of a compressor of reduced noise type which is described later. However, since the change in

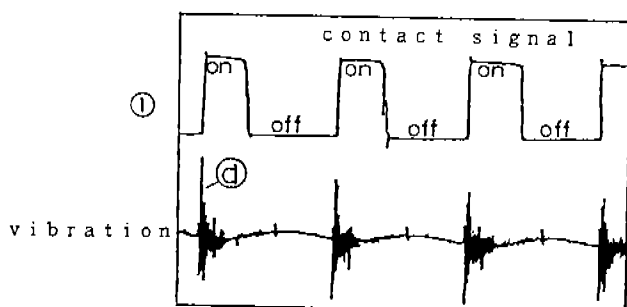
this impulsive noise that accompanies the level changes of impact waveforms (b) and (c) is only of the order which is audible to an intent ear, its contribution to the compressor noise is comparatively small.

2.4.3 Slap between the nose of the blade and the piston

Fig. 2.12 shows the results of measurements of the contact state between the blade nose and the piston.



(a) Under the test conditions



(b) At starting

Fig. 2.12 Typical data of the contact state between the blade nose and the piston under the test conditions and at starting

It will be apparent from these data that under the test conditions the nose of the blade remains in contact with the piston at all times (See Fig. 2.12(a)). At starting, however, the blade cannot follow the rotation of the piston so that a violent slap occurs repeatedly. This condition is shown in Fig. 2.12(b). Thus, an impact waveform (d) is produced, at the moment of slap, in the shell vibration acceleration waveform, whereby a large noise over 2000 Hz is produced.

While this condition persists only for a brief moment, its possible cause is that the back pressure against the blade from the shell inner space is not high enough to ensue a sufficient supply of lubricating oil.

2.4.4 Slide between the piston and the bearing

Fig. 2.13 shows some of the measured data on the change of contact between the piston end plate and the bearing.

While the piston rotates within the cylinder chamber defined between the bearings, the change in the state of slide between the piston and the bearing was investigated using the experimental device shown in Fig. 2.2(d).

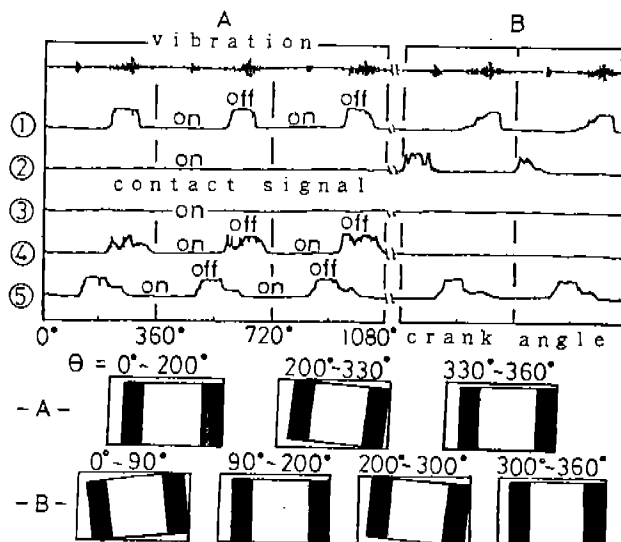


Fig. 2.13 Typical data of piston motion and shell vibration acceleration waveform, and schematic views of piston motion

The electrode attached to the bearing had been divided into 4 segments (segmental electrodes 1 to 4) so that the inclination of the piston could be monitored. The data generated by this investigation revealed that at regular cyclic intervals the piston alternately comes into the rotational state indicated by -A- in the schematic illustration in Fig. 2.13 and the rotational state indicated by -B- in the same illustration. It was found that this change in contact state between the piston and the bearing (the moment when the state of the piston undergoes a transition from an inclined position to a non-inclined position or vice versa) does not produce any definite impact waveform in the shell vibration acceleration waveform.

However, with an intent ear, a change of frictional noise could be detected in synchronism with the change of rotational position of the piston from -A- to -B-. Moreover, the rotational state of the piston varied fairly much from one test compressor to another and was not always in agreement with the data given in Fig. 2.13. Thus, no generalization could be made in this respect. It appeared that the frictional noise contributes to the compressor noise mostly at 1600 to 2000 Hz. (The noise of 1600 to 2000 Hz varies in tune with the long-cycle change in slide.)

The contact between the piston and the cylinder wall was also investigated but it was found that the piston did not contact the cylinder wall under the test conditions. However, in the case of a test compressor which was intentionally

assembled to cause a contact, a frictional noise synchronized with a long-cycle change of contact time could be heard. Thus, it was thought that the frictional noise was generated in the slide position between the piston, bearing and crankshaft. As a whole, whereas the slap and slide between the mechanical parts of the compressor give rise to impulse noise and frictional noise, respectively, their contributions are fairly small as compared with the noise due to the pressure pulsation of refrigerant gas.

3. NOISE REDUCTION MEASURES

3.1 Noise reduction measure 1 against pressure pulsation

It was stated in Chapter 2 that the noise due to the pressure pulsation in various spaces within the compressor is the greatest contributor to the noise radiated by the compressor. A first measure taken against this pressure pulsation noise was to provide a pressure pulsation buffer means in the neighborhood of the discharge port which is situated upstreams of all other positions where pressure pulsation occurs. A schematic view of this pressure pulsation buffer means is shown in Fig. 3.1.

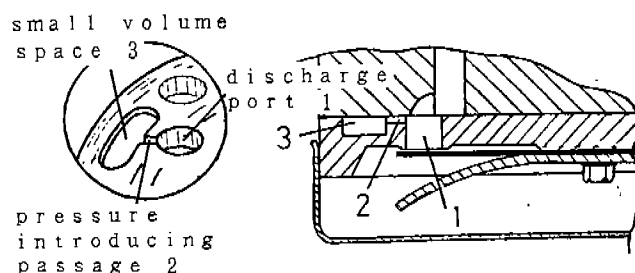


Fig. 3.1 Schematic view of the pressure pulsation buffer means provided at the discharge port

This pressure pulsation buffer means includes a small-volume space 3 communicating with the discharge port 1 of the bearing through a pressure introducing passage 2. In appearance, it is a Helmholtz resonator. Fig. 3.2 shows the pressure pulsation in the cylinder compression space (space C).

As results of the investigation indicated, the high frequency pressure pulsation indicated by mark | as observed before provision of the buffer means was no longer observed after its provision.

Fig. 3.3 shows the pressure pulsation power spectra and shell vibration acceleration waveforms before and after provision of the pressure pulsation buffer means. It is seen that in the compressor provided with the buffer the pressure pulsation is dampened over a broad frequency range, particularly in the region of 2000 to 3000 Hz. It was also found that the level of shell vibration acceleration is considerably depressed over the entire stroke. These results indicate that the very power level of the noise excited in the neighborhood of the discharge port is markedly depressed. It was, thus, confirmed that ultimately the compressor noise (shown in Fig.

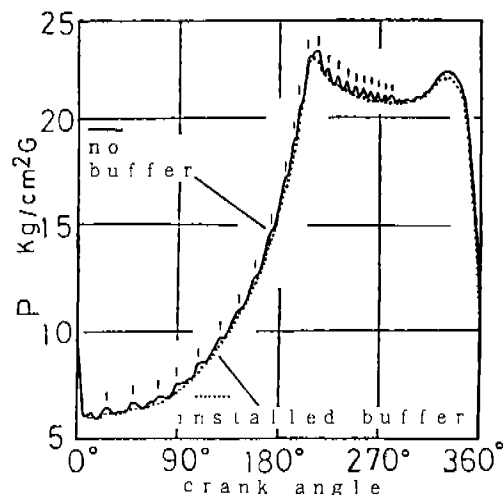


Fig. 3.2 Typical waveforms of pressure pulsation in cylinder compression space before and after provision of the pressure pulsation buffer means

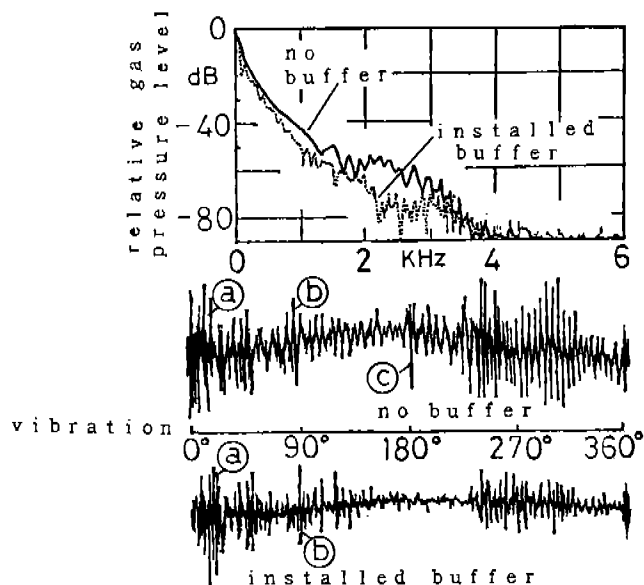


Fig. 3.3 The power spectra of cylinder compression space pressure pulsation and shell vibration acceleration before and after provision of the pressure pulsation buffer means

3.4) is significantly improved. The noise at 2500 Hz represents a contribution of accumulator noise.

In the above buffer construction, if the volume of the small-volume space 3 is increased, the noise-reducing effect is also increased proportionally but since it results in an increased top clearance, the increase of volume should be limited to a certain extent. Moreover, the noise-reducing effect varies considerably according to the dimensions of the aforementioned pressure introducing passage 2 and this suggests

that there must be an optimum value for the geometry of these elements.

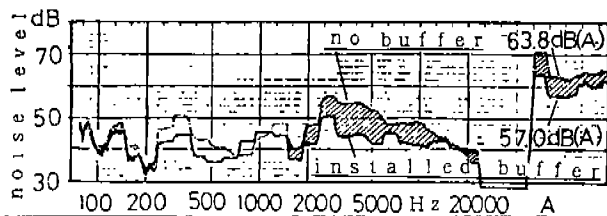


Fig. 3.4 Compressor noises before and after provision of the pressure pulsation buffer means

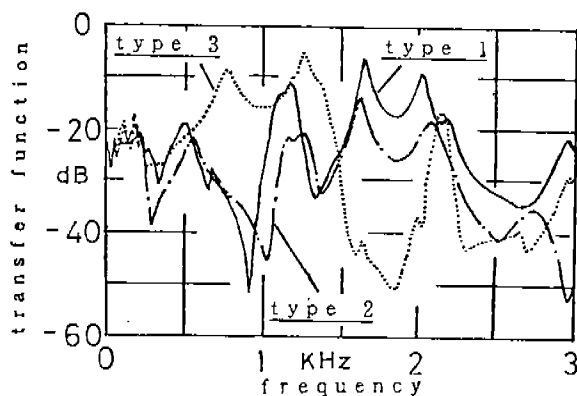
The effect of the Helmholtz resonator may be cited as a mechanism of noise reduction by this pressure pulsation buffer construction. However, there may also be contemplated a certain mechanism which reduces the rapid expansion of refrigerant gas in the neighborhood of the discharge port.

3.2 Noise reduction measure 2 against pressure pulsation

The pressure pulsation noise-reducing measure 1 described in the preceding section is remarkably effective particularly against the high frequency noise above 2000 Hz but does not produce a significant effect against the noise in the range of 500 to 1600 Hz. In this frequency range, the pressure pulsation in the shell space and discharge muffler space is a major contributor to the compressor noise, and a separate additional noise reduction measure must be provided to deal with this type of noise.

A first procedure is to optimize the discharge muffler design so as to reduce the noise radiated from the muffler exit in the aforementioned frequency range of 500 to 1600 Hz.

Fig. 3.5 shows the acoustic characteristics of various discharge mufflers experimentally fabricated.



- type 1: $V=V_0$ (space volume)
- type 2: $V=2V_0$
- type 3: discharge port, valve, and muffler installed main bearing

Fig. 3.5 Acoustic characteristics of various discharge mufflers

In conducting a comparative experiment, the compression mechanism alone was taken out from the compressor and the test was conducted in a free field. While the acoustic characteristics of shell space are shown in Fig. 2.5, it had already been confirmed by various experiments that the resonance frequency in the frequency range of 600 to 1000 Hz is a particularly important contributor to the noise. Therefore, an attempt was made to reduce the noise radiated from the discharge muffler exist in this frequency range.

From the above point of view, the discharge muffler constructions of Type 1 and Type 2 were found to be significantly superior to the construction of Type 3. Thus, in the frequency range of 500 to 1600 Hz, the discharge muffler construction should be such that the resonance frequency in the shell space will not agree with the resonance frequency in the discharge muffler space.

A second procedure is the procedure described in Reference (4). This procedure utilizes the characteristics of the first natural mode of noise excited in the shell space. Depending on its phase interference effect, the noise in the frequency range of 600 to 1000 Hz can be remarkably reduced by the procedure.

4. CONCLUSION

The results of the present experimental study may be summarized as follows.

(1) The predominant source of the noise radiated by the compressor is the pressure pulsation generated and acoustically amplified in the spaces within compressor.

(2) The pressure pulsation in the shell space and discharge muffler space has the property of air-borne noise. It is especially a dominant source of noise in the frequency range of 500 to 1600 Hz. The high frequency pressure pulsation excited in the neighborhood of the discharge port is propagated both upstreams and downstreams, and is a major source of noise over 2000 Hz.

(3) The noise due to slap and slide between the compressor mechanical parts is a noise of frequencies over 1600 Hz but contributes only a little to the compressor noise.

(4) The impulsive noise due to slap between the compressor mechanical parts consists mainly of the noise of slap between the discharge valve and seat and that between the blade and slot. Slide was observed between the piston and the bearing and between the crankshaft and the bearing.

(5) The effective measures for reducing the compressor noise consists of the provision of a pressure pulsation buffer means at the discharge port and the optimization of discharge muffler design. These procedures result in a remarkable reduction of compressor noise without affecting the compressor efficiency to any significant extent.

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